AFRL-RQ-WP-TP-2013-0236

REFRIGERANT CHARGE MANAGEMENT AND CONTROL FOR NEXT-GENERATION AIRCRAFT VAPOR COMPRESSION SYSTEMS (POSTPRINT)

Anthony Puntel, Travis Michalak, Larry Byrd, and Thomas Reitz

Mechanical and Thermal Systems Branch
Power and Control Division

Stephen Emo, Jamie Ervin, and Victor Tsao

University of Dayton Research Institute

SEPTEMBER 2013

AIR FORCE RESEARCH LABORATORY
AEROSPACE SYSTEMS DIRECTORATE
WRIGHT-PATTERSON AIR FORCE BASE, OH 45433-7542
AIR FORCE MATERIEL COMMAND
UNITED STATES AIR FORCE

NOTICE AND SIGNATURE PAGE

AFRL-RQ-WP-TP-2013-0236 HAS BEEN REVIEWED AND IS APPROVED FOR PUBLICATION IN ACCORDANCE WITH ASSIGNED DISTRIBUTION STATEMENT.

*//Signature//
TRAVIS E. MICHALAK
Program Manager
Mechanical and Thermal Systems Branch
Power and Control Division

//Signature//
THOMAS L. REITZ, Technical Advisor
Mechanical and Thermal Systems Branch
Power and Control Division
Aerospace Systems Directorate

//Signature//
JOHN G. NAIRUS, Chief Engineer
Power and Control Division
Aerospace Systems Directorate

*Disseminated copies will show "//Signature//" stamped or typed above the signature blocks.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. REPORT DATE *(DD-MM-YY)*	2. REPORT TYPE	3. DATES COVERED *(From - To)*
September 2013	Conference Paper Postprint	01 March 2012 – 26 March 2013

4. TITLE AND SUBTITLE
REFRIGERANT CHARGE MANAGEMENT AND CONTROL FOR NEXT-GENERATION AIRCRAFT VAPOR COMPRESSION SYSTEMS (POSTPRINT)

5a. CONTRACT NUMBER In-house
5b. GRANT NUMBER
5c. PROGRAM ELEMENT NUMBER 62203F

6. AUTHOR(S)
Anthony Puntel, Travis Michalak, Larry Byrd, and Thomas Reitz (AFRL/RQQM)
Stephen Emo, Jamie Ervin, and Victor Tsao (University of Dayton Research Institute)

5d. PROJECT NUMBER 3145
5e. TASK NUMBER
5f. WORK UNIT NUMBER Q0LA

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
Mechanical and Thermal Systems Branch (AFRL/RQQM)
Power and Control Division, Air Force Research Laboratory
Aerospace Systems Directorate
Wright-Patterson Air Force Base, OH 45433-7542
Air Force Materiel Command, United States Air Force

University of Dayton
Research Institute
300 College Park Avenue
Dayton, OH 45469

8. PERFORMING ORGANIZATION REPORT NUMBER
AFRL-RQ-WP-TP-2013-0236

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)
Air Force Research Laboratory
Aerospace Systems Directorate
Wright-Patterson Air Force Base, OH 45433-7542
Air Force Materiel Command
United States Air Force

10. SPONSORING/MONITORING AGENCY ACRONYM(S)
AFRL/RQQM

11. SPONSORING/MONITORING AGENCY REPORT NUMBER(S)
AFRL-RQ-WP-TP-2013-0236

12. DISTRIBUTION/AVAILABILITY STATEMENT
Approved for public release; distribution unlimited.

13. SUPPLEMENTARY NOTES
PA Case Number: 88ABW-2013-1460; Clearance Date: 26 Mar 2013. This paper contains color. The conference paper was presented at the 2013 SAE AeroTech Congress & Exhibition Conference, held in Montreal, Quebec, Canada from September 24 through 26, 2013, and was published in the Proceedings of the 2013 SAE AeroTech Congress & Exhibition.

14. ABSTRACT
Vapor compression systems (VCS) offer significant benefits as the backbone for next generation aircraft thermal management systems (TMS). For a comparable lift, VCS offer higher system efficiencies, improved load temperature control, and lower transport losses than conventional air cycle systems. However, broad proliferation of VCS for many aircraft applications has been limited primarily due to maintenance and reliability concerns. In an attempt to address these and other VCS system control issues, the Air Force Research Laboratory has established a Vapor Cycle System Research Facility (VCSRF) to explore the practical application of dynamic VCS control methods for next-generation, military aircraft TMS.

The total refrigerant mass contained within the closed refrigeration system (refrigerant charge) is a critical parameter to VCS operational readiness. Too much or too little refrigerant can be detrimental to system performance. Extreme values of refrigerant charge can lead to a loss of evaporator temperature control, loss of high side pressure control, or other potentially catastrophic occurrences. The objective of this work is to examine real-time methods for determination of acceptable refrigerant charge in a prototypical VCS system, as a function of operational points, using only sensors already utilized in the control system (in situ control sensors). It is envisioned that studies such as these can be used to guide development of a simple in situ prognostic tool for system state-of-health indication (i.e., “Red Light, Yellow Light, Green Light”), with respect to level of charge, and to enable on-demand maintenance. Additionally, a method for continuous management of refrigerant charge as a means for optimizing system efficiency over a range of dynamic operating points is presented.

15. SUBJECT TERMS
VCS, Vapor Cycle System, VCS Control, Refrigerant Charge Level, Aircraft Thermal Management

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT:	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON (Monitor)
Unclassified	Unclassified	Unclassified	SAR	14	Travis E. Michalak **19b. TELEPHONE NUMBER** *(Include Area Code)* N/A

2013-01-2241

Published 09/17/2013, doi:10.4271/2013-01-2241

Refrigerant Charge Management and Control for Next-Generation Aircraft Vapor Compression Systems

Anthony Puntel
US Air Force Research Laboratory

Stephen Emo
UDRI

Travis E. Michalak
US Air Force Research Laboratory

Jamie Ervin
UDRI

Larry Byrd
US Air Force Research Laboratory

Victor Tsao
UDRI

Thomas Reitz
US Air Force

ABSTRACT

Vapor compression systems (VCS) offer significant benefits as the backbone for next generation aircraft thermal management systems (TMS). For a comparable lift, VCS offer higher system efficiencies, improved load temperature control, and lower transport losses than conventional air cycle systems. However, broad proliferation of VCS for many aircraft applications has been limited primarily due to maintenance and reliability concerns. In an attempt to address these and other VCS system control issues, the Air Force Research Laboratory has established a Vapor Cycle System Research Facility (VCSRF) to explore the practical application of dynamic VCS control methods for next-generation, military aircraft TMS.

The total refrigerant mass contained within the closed refrigeration system (refrigerant charge) is a critical parameter to VCS operational readiness. Too much or too little refrigerant can be detrimental to system performance. Extreme values of refrigerant charge can lead to a loss of evaporator temperature control, loss of high side pressure control, or other potentially catastrophic occurrences. The objective of this work is to examine real-time methods for determination of acceptable refrigerant charge in a prototypical VCS system, as a function of operational points, using only sensors already utilized in the control system (in-situ control sensors). It is envisioned that studies such as these can be used to guide development of a simple in-situ prognostic tool for system state-of-health indication (i.e. “Red Light, Yellow Light, Green Light”), with respect to level of charge, and to enable on-demand maintenance. Additionally, a method for continuous management of refrigerant charge as a means for optimizing system efficiency over a range of dynamic operating points is presented.

INTRODUCTION

Advanced electronic packages are challenging aircraft thermal management systems (TMS) in terms of higher cooling loads. This trend is forecast to continue for the foreseeable future. This trend is driven by more-electric architectures with higher electrical power (and cooling) requirements, bleed-less engine architectures, and higher power directed energy devices. To complicate this increase in load, there has also been a decrease in available aircraft heat sinks, especially low-temperature sinks. Traditionally, engineers have relied upon fuel, ram air, or engine bypass air as heat sink for the TMS coolant flows. In many military applications these sinks are diminishing due to engine efficiency improvements, stealth implications, and efforts to employ more bleed-less engine architectures. Advanced three-stream engine architectures, such as that under development through the Adaptive Versatile ENgine Technology (ADVENT) program, partially ameliorate this deficit, though the availability of this sink depends upon the aircraft's position in the flight envelope. As such, there is a need for a dynamically responsive TMS which can intelligently respond to changing sink conditions to manage the thermal requirements of the aircraft. These changes are forcing new thermal management architectures. Prior generation aircraft have almost exclusively used air cycle systems (ACS). These systems were primarily engine bleed air driven. The ACS system does not depend on a special fluid and can reject heat at much higher temperatures with moderate pressures compared to a vapor compression system (VCS) with conventional refrigerant. However, ACS consume more energy (up to 10X) and require far greater volumetric flows compared to VCS for the same temperature lift. However, system controllability concerns, limited maximum sink temperatures, and refrigerant leakage concerns have restricted the broad proliferation of VCS on-board aircraft [1].

VCS were invented in the 1830s, and have been in practical use since the 1850s [2]. However, their application in aviation systems has been limited. Terrestrially, a VCS can be found in almost every aspect of modern life, ranging from cars, homes, supermarkets, large buildings, processing plants, and computer room cooling. In most of these applications, the loads and sinks vary relatively slowly, primarily by diurnal cycling. As a specific example, a building sinking temperature will vary throughout the diurnal cycle and with seasonal changes. Nearly all of these changes occur on the scale of hours, thus appearing relatively steady-state. Likewise, VCS currently used on-board aircraft such as the E2C, F-22, A-380, and Apache-Longbow are fundamentally steady-state applications, wherein their platform loads do not vary significantly over the flight envelope. Next generation aircraft are anticipated to have far more demanding variations in terms of both load and sink changes. The challenge to the systems designer is to accommodate not only large dynamic swings in load (turn down ratio), but also major swings in both the sink temperature and the availability of the various heat sinks. Additionally, these future systems can potentially have changes that occur in seconds rather than hours.

Not only do the controls for these new VCS need to be much more responsive than those for typical ground based systems, but they need to meet flight critical failure modes and detection standards. Unlike an ACS, which is generally an open loop or semi-closed loop system, a VCS is operated closed loop. A VCS requires a minimum mass of refrigerant (charge) to maintain cooling performance [3]. As VCS begin to take over cooling duties of flight critical loads traditionally served by ACS, concerns arise about the variation in TMS performance as a function of refrigerant charge, proper measurement of refrigerant charge, and minimally intrusive maintenance methods to sustain potentially flight-critical operation. It has been shown previously that VCS performance (controllability and efficiency) can be influenced by refrigerant charge. However, questions remain regarding proper refrigerant charge to promote optimal system operation in view of potentially large and relatively rapid variations in sink temperatures.

For a minimalistic aircraft VCS health prognostic strategy, it is desirable to be able to determine at least 4 charge states: excessive charge, acceptable charge, low but acceptable charge, and insufficient charge. It is generally accepted that excessive charge results in higher than desirable Saturated Discharge Temperatures (SDT) or high side pressure. Likewise, it is also generally accepted that low charge can result in loss of potential subcooling resulting in greater compressor flow demands and eventual loss of load temperature control.

The objective of this paper is to explore the relationship between system refrigerant charge and performance under a range of operating conditions. The results are presented from a series of tests conducted at the Air Force Research Laboratory's VCSRF. A method of relative charge detection and dynamic charge control to ensure optimal VCS performance is also proposed.

EXPERIMENT DESCRIPTION

Figure 1 shows a schematic of the VCSRF system used in this work. The hardware was the same as that previously reported, which includes a variable speed screw compressor from Fairchild Controls Corporation, a Danfoss 70kW condenser (B3-095-72-H), two Emerson expansion valves (EX-4), and two parallel Fairchild Controls Corporation 18kW evaporators (previous paper stated 12kW, which was in error) [1]. The evaporators were individually heated with MIL-F-5606 oil from two independently variable 0-12kW inline heaters. The condenser was cooled with a 75% / 25% propylene glycol / water mixture from a 60kW facility chiller. The chiller outlet temperature and flow could be independently varied from 20 to 100 °F and 0 to 50GPM, respectively. For the purposes of this experiment, several changes were made to this system to improve system responsiveness and control including:

- All receivers were bypassed
- All filter/driers were bypassed
- Control architecture was altered from superheat/capacity to cycle-optimizing.
- Compressor motor drive used was a Yaskawa A-1000.

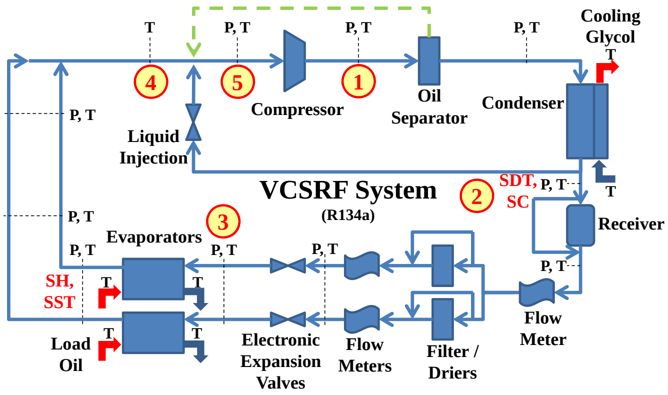

Figure 1. VCSRF Schematic Diagram

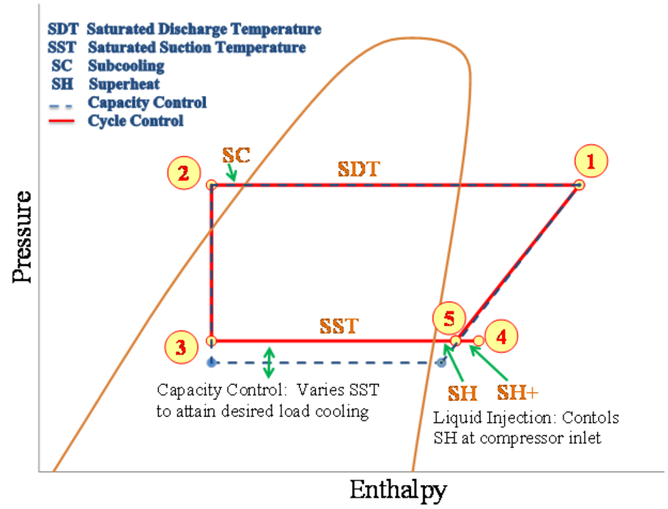

Figure 2. Cycle Diagram

The control architecture was significantly altered. In the original paper [1] the controls mimicked a traditional capacity and superheat based control system, although it used electronics expansion valves (EXV) in lieu of the traditional thermal expansion valves (TXV). Figure 2 shows an idealized thermodynamic cycle (on a Pressure-Enthalpy diagram) for the system depicted in Figure 1, with the state points on Figure 2 corresponding to the numbered locations on Figure 1. Figure 2 shows an exaggerated difference between the two control cycles. The dashed blue lines show the original superheat and capacity control method, and solid orange lines show the cycle-optimizing based system. The original architecture used the EXVs to control the refrigerant evaporator discharge superheat (SH) (emulating traditional TXV control) in conjunction with modulating the compressor speed to control the evaporator load discharge temperature. This control scheme works very well for loads that have similar set points. In the VCSRF, precautions were taken to ensure that the compressor inlet remained within the manufacturer specified acceptable inlet SH range by using a liquid injection system. This allowed the evaporator SH to be operated independent of the compressor inlet SH. VCSRF also had a closed loop control of the SDT by varying the condenser coolant flow; this was common to both control approaches.

The new architecture retained the condenser SDT control but switched to a cycle-optimizing based control for the evaporator EXVs and compressor speed. One goal is to operate the VCS system with the minimum power; to do this it is desirable to run at the lowest high side pressure, and the highest low side pressure, to minimize the pressure rise across the compressor. In this control approach, the compressor speed was modulated to hold the saturated suction temperature (SST) at a fixed temperature difference below the coldest evaporator temperature set point. In this case, the evaporator temperature set point refers to the desired oil temperature at the exit of the evaporator. The EXV openings were then independently modulated to control the oil temperature at the outlet of the evaporator (load temperature) to their respective set points. The refrigerant SH at the exit of the evaporator was not directly controlled in this control scheme. The primary benefits of this approach are: minimizing pressure rise across the compressor, enabling independent control of parallel evaporator(s) without the use of backpressure control valve(s), and decoupling the evaporators from the compressor capacity control. The liquid injection loop is also active at all times, to maintain control of the SH of the vapor at the compressor inlet in either control approach. Referring to Figure 2, the liquid injection contribution is shown as the enthalpy and temperature decrease from state 4 to 5.

Table 1 lists the estimated system volumes used in the current experiments. Note that the elimination of the receiver and filter driers reduced the high side volume significantly, resulting in a charge reduction of greater than 50%. In comparison with previous work by the authors, the current configuration reduces refrigerant charge by 27 lbs. It should be noted that although the high side and low side volumes are nearly equivalent, the mass distribution between these sections is substantially different at 20:1, respectively. Throughout this test it was assumed that the mass on low side is essentially constant. Therefore changes in refrigerant mass approximate the change in condenser liquid level once the downstream high side pipes to the EXV are filled.

Table 1. System Volumes and Relative Mass Distribution

Component	Vol (ft^3)	%Vol	Mass (lb_m) SDT = 125 °F
LP Piping	0.278	20.8	0.291
HP Piping	0.195	14.6	13.122
Condenser	0.254	19.0	6.294
Evaporator	0.073	5.5	0.708
Oil Separator	0.121	9.1	0.480
Compressor	0.047	3.5	0.121
total	0.968	100	21.016

The accuracies and ranges of the sensor suites used for these tests are shown in Table 2.

Table 2. Sensor Accuracy and Range

Instrument	Accuracy	Range
High Pressure-Pressure Transducers	+/- 0.76% FS	0-500 psia
Low Pressure-Pressure Transducers	+/- 0.33% FS	0-150 psia
Glycol and Hydraulic Oil Press Transducers	+/- 0.16% FS	0-150 psia
Thermocouples	+/- 0.15 °F	10-250 °F
Glycol Flow Meter	+/- 0.25% FS	15-50 GPM
Hydraulic Oil Flow Meter	+/- 0.11% FS	0-20 GPM
Refrigerant Flow Meter	+/- 1% FS	0.5-5 GPM

EXPERIMENTAL PROCEDURES

Independent variables for this experiment included refrigerant mass charge, individual heater power, evaporator oil flow rate, SDT set point, condenser cooling fluid inlet temperature, and the evaporator oil discharge set points. The focus of these tests was to assess the impact of refrigerant charge on VCS efficiency (coefficient of performance, (COP)) and ability to maintain evaporator load temperature within ±2 °F of the set point. To accomplish this, the refrigerant charge was systematically changed and system performance was characterized. The refrigerant charge was varied gravimetrically through both high and low side sections of the VCSRF. Typical operation proceeded as follows. The VCSRF was started and allowed to stabilize at constant evaporator loads of 12kW, each, with fixed 65 °F evaporator oil temperature set points; refrigerant charge was then slowly added or removed from the system in 1 or 2 pound increments and allowed to stabilize, at which point system performance was characterized. This was repeated at three SDT settings and up to 6 different condenser inlet coolant temperatures. In this manner it was possible to examine the relationships of charge to COP, compressor speed, high side pressure, condenser refrigerant exit temperature, subcooling, expansion valve position, and the evaporator outlet load temperature.

RESULTS AND DISCUSSION

The following sections explore the relationship between refrigerant charge parameters, SDT, and condenser sink temperature on system COP and load temperature controllability. The following sections will be described in terms of SDT control. The SDT control is not actually a temperature control loop. The SDT control loop is actually a pressure control loop. This pressure is also called the high-side pressure. Referring back to Figure 2, it is the pressure at the top of the cycle between points 1 and 2. Likewise, when SST control is discussed, it is also a pressure control loop. This pressure is also referred to as the low-side pressure, which is the pressure depicted in Figure 2 between points 3 and 4.

COP AS A FUNCTION OF CHARGE AT FIXED SDT AND SINK TEMPERATURES

It is generally accepted that increasing subcooling increases the COP to a limit, at which point the system will experience an unwanted increase in high side pressure due to excessive condenser flooding. This increase in COP is the result of an increase in available enthalpy change across the evaporator(s), which decreases the required mass flow and thus reduces the compressor speed needed to maintain the evaporator load temperature(s). This increase in available enthalpy change would be seen in Figure 2 as points 2 and 3 moving to the left, corresponding to the increase in subcooling and the decrease in evaporator inlet enthalpy.

Figure 3 shows the relationship, as measured on the VCSRF, between condenser discharge temperature and COP with a constant load of 12kW on both evaporators, an SDT set point of 135 °F, and a condenser inlet sink temperature of 80 °F.

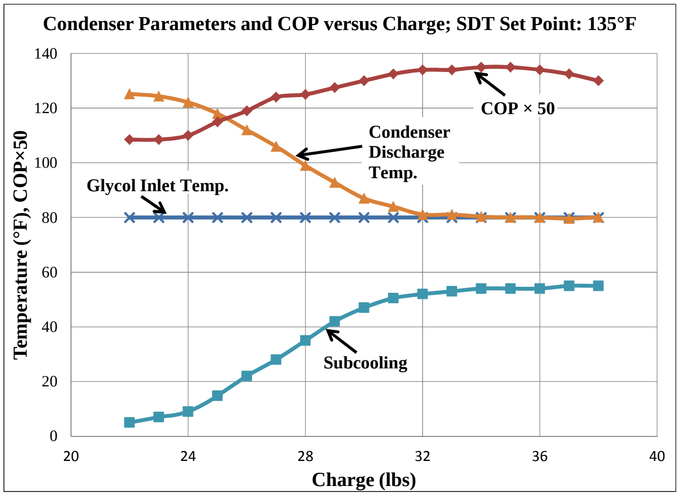

Figure 3. Subcooling and Charge

Subcooling is defined as the difference between the condenser refrigerant exit temperature and condensing temperature, ideally the SDT set point. As shown in Fig. 3, as the refrigerant charge was increased, the condenser discharge temperature decreases to the limit of the sink temperature, from 22 to 32 pounds of charge. Throughout this range of charge the subcooling likewise is increasing. Above 32 pounds of charge, there is negligible change in condenser discharge temperature or subcooling. The COP increases from 22 to 32 pounds of charge and then remains relatively constant from 32 to 36 pounds. Above 36 pounds of charge the COP begins to decrease and the subcooling appears to increase. This would indicate an increase in high side pressure and inability to maintain SDT or high side pressure.

Figure 4 examines the impact on compressor speed, and corroborates a reduction in speed as the subcooling increases. This corresponds directly with the increase in available enthalpy change as the subcooling increases.

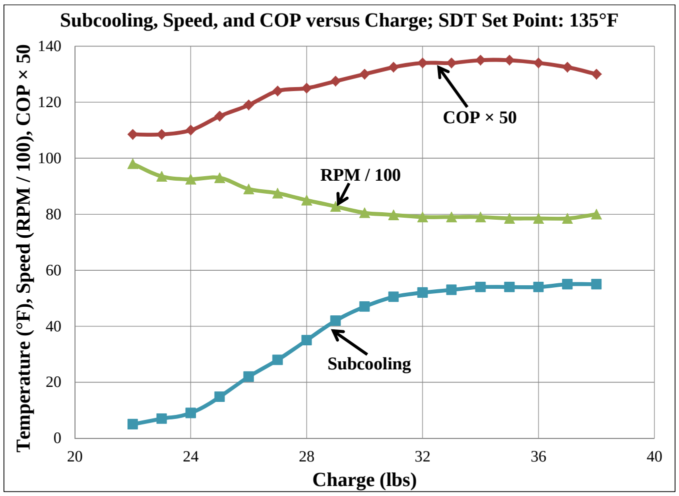

Figure 4. COP and Compressor Speed

Referring back to Figure 3 there was a decrease in COP above 36 pounds of charge. Figure 5 brings the high side pressure and condenser coolant flow into consideration. Figure 5 shows an increase in high side pressure above 36 pounds of charge. This increase in pressure occurs at the same time as a sudden large increase in condenser coolant flow. Not shown in the data is that above ~37 pounds of charge, the SDT control loop commanded the coolant flow to maximum flow. This is a direct result of the control loop no longer being able to contain the high side pressure. The condenser heat transfer is limited by one of three factors: the heat capacity of the coolant ($\dot{m}C_p$), phase change enthalpy of the refrigerant ($\dot{m}h_{fg}$), or product of the condensing area (A) and overall heat transfer coefficient (U) of the heat exchanger itself. Figure 5 shows that the condensing capacity is sensitive to coolant flow from 22 to 36 pounds of charge. In this region the limiting factor in the condenser heat transfer was the heat capacity of the coolant ($\dot{m}C_p$). Above 36 pounds, the limiting factor must be the condensing area A. The decrease in available condensing area is due to more and more of the condenser volume being filled with liquid refrigerant, as the additional mass of refrigerant added to the system will collect in the condenser as liquid. This insufficient condensing area results in the loss of SDT control which appears as both an increase in SDT and high side pressure. The increase in high side pressure results in an increase in compressor pressure rise and thus an increase in work, resulting in a reduction in COP.

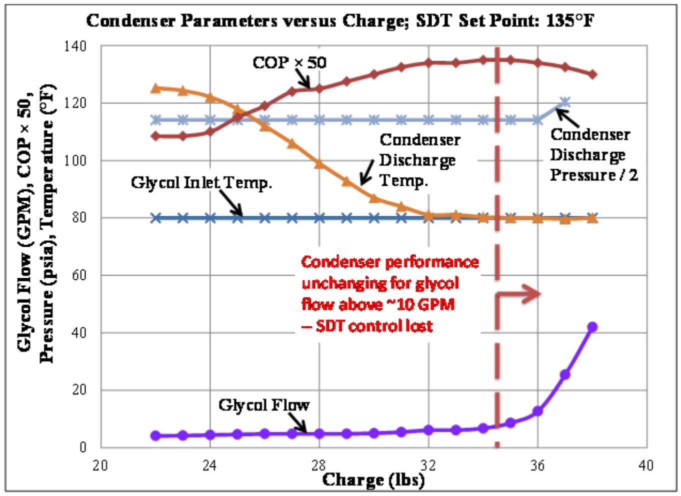

Figure 5. Condenser Discharge Pressure and Sink Flow

Figures 3-5 summarize the effect of charge on COP and what occurs with excessive charge. Figure 6 examines the effect of too little charge. This figure clearly shows the system lost the ability to control Panel 4 evaporator oil temperature below ~23 pounds of charge. Even though the graph shows a positive subcooling, there was vapor visible in the main line and Panel 4 sight glasses below 24 pounds of charge. At the time of writing, it is unclear what causes the difference between the subcooling calculation and the visual observation of vapor in the line. In addition, it was observed that once the charge dropped below 24 pounds, the compressor speed became more erratic. This correlates with the visual vapor observation at the entrance of the Panel 4 EXV. Even though the system was able to maintain the load temperature control it was becoming marginal. From Figure 6, one could infer, in the case of VCSRF, that the system had insufficient charge when the EXV position was greater than ~50%. This parameter can provide clear indication of too little charge and incipient loss of control. Admittedly this value is dependent on the EXV size selected and the load. In the VCSRF case the EXV are oversized for the load. For instance, during these tests, the load was at 66% of rating while the EXV were only 25% open.

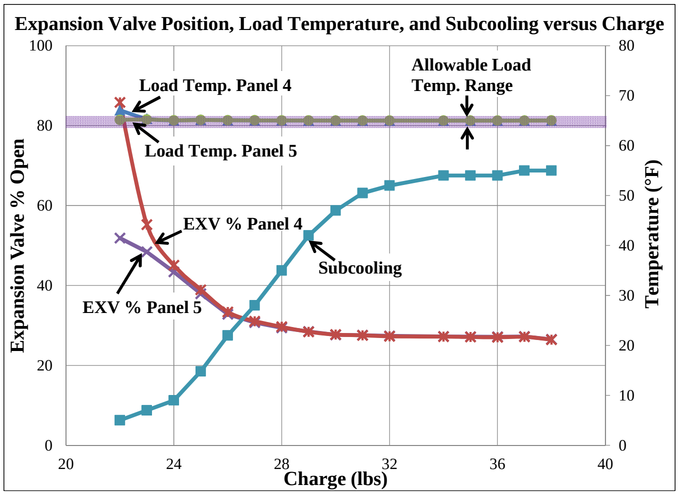

Figure 6. Expansion Valves and Low Charge

Figure 6 also shows that Panel 5 remained within regulation as low as 21 pounds. Although the EXVs and evaporators used on Panels 4 and 5 are identical, the plumbing connecting the components was different. Figure 7 illustrates schematically the differences in the plumbing.

For the data in Figures 3-6, the plumbing was in configuration 1, as shown in Figure 7a. Valves V1 and V4 were both closed and V2 and V3 were open. For the data in Figures 8 and 9, the plumbing was in configuration 2, as shown in Figure 7b, with valves V2 and V4 closed, and valves V1 and V3 open. In both configurations, the refrigerant coming from the condenser first travelled through the lower line, at an elevation approximately 3 feet below the evaporator EXVs. It had to travel vertically up, either through the riser containing valve V2 or the riser at the far right-hand side of the schematic, to reach the EXVs.

As was mentioned previously, at low charges the system was running without subcooling, as evident in the system behavior and confirmed using sight glasses. When the system was running in configuration 1 (Figure 7a), EXV 4 was fed first. At low charge levels (without subcooling), the two-phase fluid passed from the condenser down the low line to the first riser, at which point the vapor preferentially went up the vertical riser, which effectively separated the vapor from the liquid. This vapor was fed to EXV 4, allowing the liquid to continue on to Panel 5. Vapor was never observed in the sight glass on Panel 5 when the system was running in configuration 1. Thus, in this configuration, Panel 4 acted as a vapor separator for Panel 5.

When the system was running in configuration 2, all of the refrigerant passed under the evaporators, to go up the riser at the far right-hand side of the schematic. In this configuration, Panel 5 is fed first. In this case, however, the liquid would preferentially fall down the supply line feeding Panel 5, again allowing vapor to be fed to Panel 4. Thus, in either configuration, Panel 4 tended to lose control of the load temperature first. However, in configuration 2, the system was able to continue operation down to 17 pounds, whereas the minimum charge level in configuration 1 was approximately 23 pounds.

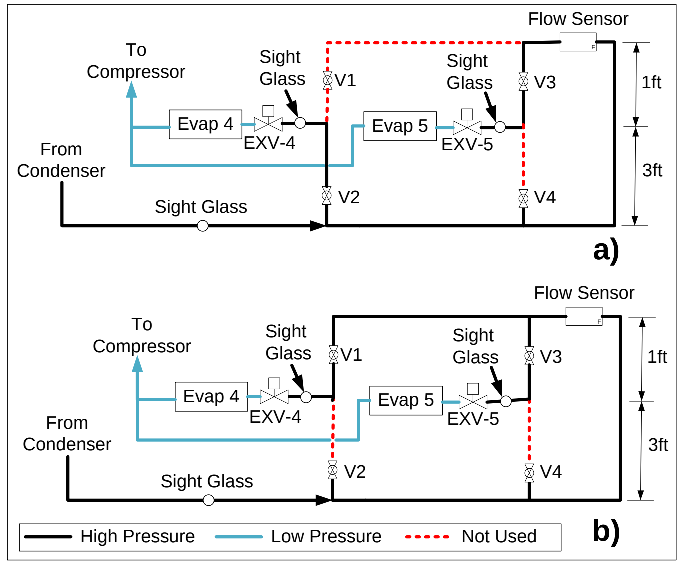

Figure 7. Pipe Schematic, Elevation View:
a) Configuration 1, b) Configuration 2

By combining Figures 3 through 6, one can conclude that VCSRF system will stay within regulation between 23 and 36 pounds of charge. In practice, it is impractical to know the absolute charge without physically evacuating it and recharging the system. It is therefore desirable to discuss charge management in terms of either addition or subtraction of charge. The leading signs of low charge are high % EXV opening and loss of subcooling. Low subcooling can also be attributed to low temperature difference between SDT and the condenser sink temperature rather than low charge.

The effect of system charge and sink temperature at a constant SDT on the VCSRF COP and the operability is shown in Figure 8. The vertical red dashed line represents the minimum charge below which load temperature control was lost. The blue dashed line represents the boundary beyond which SDT control was lost. The remaining 3 traces show the measured COP values as a function of charge at a constant SDT set point of 135 °F but at 3 condenser sink temperatures. To the right of the blue dashed line SDT control is lost and there is a very significant drop in COP. It appears that the drop in COP becomes more acute as the charge increases, which supports the loss of condensing area with increasing charge.

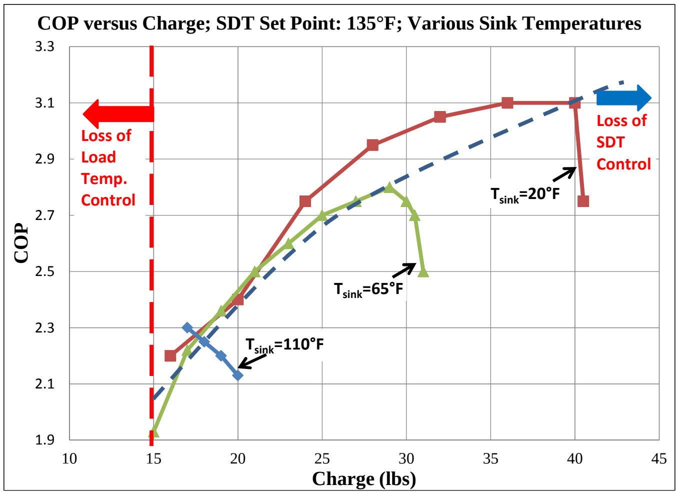

Figure 8. COP with Sink Temperature and Charge.

Figure 8 also shows that the allowable charge range varies with the differences between SDT and the sink temperatures. At small temperature differences, such as with a 110 °F sink temperature, there is only a 3-pound window (16 to 19 pounds) of charge over which the system can be safely operated, whereas at a 20 °F sink temperature (large temperature difference) the charge range is nearly 24 pounds. If the VCSRF system had to operate with a fixed charge over this range of sink temperatures (20 to 110 °F), with this fixed SDT, the fixed charge would need to be between 16 to19 pounds, which corresponds to the most restrictive case of a 110 °F sink temperature. The resulting COP would be, at best, around 2.3. If the charge could be varied as a function of sink temperature, for instance, charging the system to 36 pounds when the sink temperature is 20 °F, there is a potential 34% improvement in COP.

Figure 9 is a summary plot showing the values of peak COP for SDT set points and condenser sink temperatures. Additionally the dashed overlay line is the SDT for the 20 °F sink temperature and 135 °F SDT from Figure 8.

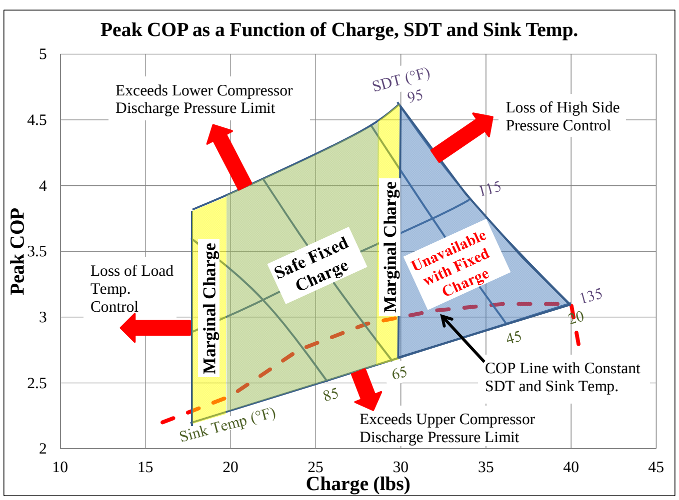

Figure 9. Peak COP Map for VCSRF

Figure 9 depicts the safe charge range for VCSRF with the assumed range of sink and SDT temperatures. Below 17 pounds of charge, load temperature control is lost. Vapor begins to appear in the main condenser exit line at 25 pounds or less, but due to the change in routing the system can maintain control to 17 pounds. In this case, safe and stable operation was possible without any subcooling. The far right line represents the maximum charge allowable for a 20 °F sink temperature over the range of SDTs. Above 30 pounds of charge, not all combinations of SDT and sink temperature are allowable. For example, 34 pounds of charge would result in excessive high side pressure at SDTs below 115 °F.

Therefore, if one had to pick a fixed charge region to allow operation of the VCSRF over the depicted SDT range of 95 to 135 °F, and sink temperatures in the range of 20 to 65 °F, it would be 17 to 30 pounds for operability. In practice one would place a 2- to 5-pound margin at each extreme, leaving a nominal charge in the 20- to 28-pound range.

ACTIVE CHARGE CONTROL SYSTEM

Controlling VCSRF to operate at maximum efficiency would dictate that the system run at the lowest SDT possible based on the available sink temperature. Figure 8 illustrates that acceptable operating charge is also a function of the temperature difference between SDT and sink temperature. This is best observed at the high sink temperature of 110 °F, where there is a narrow range of only a few pounds of charge between loss of load temperature control and loss of SDT or high side pressure control. As this temperature difference increases, the allowable charge range also increases; at a 65 °F sink temperature, the allowable charge range increases to 13 pounds.

Referring back to Figure 9: if, for example, the system is running at 115 °F SDT with a sink temperature of 65 °F, the peak COP would be ~ 3.3 with approximately 26 pounds of charge. COP could be increased to 4.1 by decreasing the SDT temperature to 95 °F, but then the charge must be reduced to around 23 pounds. If the charge were fixed at 26 pounds, the high side pressure could not be reduced. The high side pressure would be equivalent to the corresponding 115 °F SDT pressure. In this example, an active charge control system would result in a 21% increase in COP.

The sink temperature of airborne applications can vary widely, from -60 °F to over 100 °F, thus making it very advantageous to dynamically adjust the system charge to achieve peak COP at minimum SDTs.

We will explore the behavior of an active charge system in future work.

SUMMARY AND CONCLUSIONS

In this paper, it has been shown that charge affects nearly every aspect of the operation of a VCS. It has been shown that there are three distinct operating regions; too high of charge, too low of charge, and an acceptable range. These states can be detected as follows:

Low Charge: Manifests as loss of load temperature control, excessive EXV opening, and lack of subcooling. The detection of incipient failure is loss of subcooling, and excessive EXV opening exceeds ~50%. The 50% value will vary depending on the evaporator load and relative EXV capacity.

Marginal Charge: Charge can be either marginally low or marginally high. Marginally low charge manifests as EXV openings higher than nominal, but less than the 50% limit. Marginally high charge could potentially manifest as a small difference between sink and condenser exit temperature.

Adequate Charge: The system maintains SDT control and load temperature control.

High Charge: Manifests as an inability to maintain SDT control, resulting in excessive high side pressures. Note the subcooling can actually exceed the desired level when the actual SDT exceeds the SDT set point, while the refrigerant discharge temperature remains constant.

The data also suggests that it would be advantageous in terms of both operability and efficiency to have an active charge control system for future airborne VCS systems with wide ranges of sink temperatures.

The authors propose to implement an active charge system as part of the VCSRF and measure the potential savings over a number of varieties of simulated mission profiles.

REFERENCES

1. Byrd, L.W., Cole, A., Emo, S., Ervin, J. et al., "In-situ Charge Determination for Vapor Cycle Systems in Aircraft", SAE Technical Paper 2012-01-2187, 2012, doi: 10.4271/2012-01-2187 .
2. Cengel, Y.A. and Boles, M.A., "Thermodynamics: An Engineering Approach," 4th Ed., McGraw-Hill, New York, ISBN 0-07-238332-1: pg. 569, 2002.
3. Kim, W. and Braun, J.E., "Evaluation of a Virtual Refrigerant Charge Sensor," presented at International Refrigeration and Air Conditioning Conference, USA, Paper 1121, 2010, http://docs.lib.purdue.edu/iracc/1121 .
4. Lemmon, E.W., Huber, M.L., and McLinden, M.O., "NIST Reference Fluid Thermodynamic and Transport Properties – REFPROP, Version 9.0", Gaithersburg, MD, 2010.

CONTACT INFORMATION

Travis Michalak, AFRL/RQQM, 1950 5th St., Bldg. 18 Wright-Patterson AFB, OH 45433; Phone (937) 656-4429; email: travis.michalak@wpafb.af.mil

ACKNOWLEDGMENTS

The authors acknowledge the support of AFRL/RQQM Contract No. FA8650-04-D-2403 – DO17, as well as the support of Fairchild Control Corporation under CRADA 09-239-RZ-01. The authors would also like to thank Brian Cranston and Bill Melke for their valuable assistance with this effort.

DEFINITIONS/ABBREVIATIONS

ACS	air cycle system
ADVENT	Adaptive Versatile ENgine Technology
AFRL	Air Force Research Laboratory
COP	coefficient of performance
CRADA	cooperative research and development agreement
INVENT	INtegrated Vehicle ENergy Technology
SDT	saturated discharge temperature
SST	saturated suction temperature
TMS	thermal management system
UDRI	University of Dayton Research Institute
VCS	vapor cycle system